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COLD-START MINIDIESEL ENGINE DEVELOPMENT

By

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Military mission performance in a chemically contaminated environment necessitates the wearing of chemically impermeable protective garments. Soldiers encapsulated in these protective garments will succumb to heat stress in a hot environment. Consequently, a microclimate cooling system is being developed. In the design, a vapor compression cycle chills water which is then circulated over the body to absorb heat via a cooling garment. The engine, the prime mover in the system, is a major component. Commercially available engines of approximately 0.5 hp at 4000 rpm, the required size, include model aircraft engines and small outdoor power equipment (string trimmer) engines. However, these engines do not run on JP-8 or diesel fuel, a requirement for military use. This report details design, evaluation, and results of three engines. These include: the potential of a high compression, carbureted,glow plug engine operating on diesel fuel; the potential of a high compression, spark assist, fuel injected engine operating on diesel fuel; and a true high compression, fuel injected engine operating on diesel fuel.

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PREFACE

The work reported herein was performed under contract number DAAK60-91-C-0093, from July 26, 1991 to March 13, 1992.

Scott Bennet of the Individual Protection Directorate was the Natick Project Officer for this report.

COLD-START MINIDIESEL ENGINE DEVELOPMENT

Indroduction

Advanced Materials, Inc. (AMI) used three different approaches to identify the most promising technology for developing a true cold start diesel engine. The approaches are as follows:

1. Evaluation of the potential of a high compression carbureted glow plug model operating on pure diesel fuel. This involved investigations with the PAW brand altered fuel diesel engine and investigations with an O/S brand glow plug engine.

2. Evaluation of the potential of a high compression, spark assist, fuel injected system operating on pure diesel fuel.

3. Evaluation of the potential of a true high compression, fuel injected minidiesel engine operating on pure diesel fuel.

The development work and the corresponding results achieved to date for each of these approaches are presented in this report.

Carbureted Diesel Engine: Compression Ignition and Glow Plug Ignition

Two different types of model aircraft engines were experimented with to determine the feasibility of running these engines on pure diesel fuel. The main strategy employed in the conversion process was to utilize the combination of compression and catalytic action to sustain the combustion of diesel fuel. These engines were somewhat different than regular diesel engines in that they employed a carburetor instead of an injection system for fuel delivery. It was thought that the use of a carburetor instead of a fuel injection would make the engine more reliable, and simpler to operate and maintain. Before any in-depth details of the trials are discussed, some background information will be presented to support the approach taken during this phase of the project.

Background

Combustion in diesel engines is dependent on local conditions within each part of the charge, and not the spread of flame as in spark ignition engines. There are three phases of combustion in a diesel engine: the delay period, the rapid combustion of the fuel, and the combustion of the rest of the fuel. For the design of small engines, the most important factor is the delay period. The delay period is influenced by several different factors. The length of the delay period depends on the pressure and temperatures that exist in the cylinder gases, as well as the cetane rating of the fuel. The fuel/air ratio and the degree of mixing also play a role in the determination of the delay time.

For small engines the surface-to-volume ratio is so high that often it is not possible to retain the necessary heat to sustain combustion. This is because the high surface-to-volume ratio also serves to lengthen the delay time. This means that the optimum point of the cycle for ignition might have passed before the fuel can autoignite. This problem is compounded in small, high-speed diesel engines. The length of the delay time is not dependent on the speed of the engine and, therefore, as the speed of the engine is increased, the fuel must be injected into the cylinder earlier in the cycle to compensate for this factor. These two factors together point to a definite limit on how small an engine can be made using conventional diesel technology.

As mentioned earlier, the delay time can be influenced by the cetane rating of the fuel. Diesel fuels are a mixture of petroleum fractions lying between kerosene and lubricating oils. The main components of the fuel are paraffins, isoparaffins, olefins, napthene and aromatics (benzene). These compounds are listed in order from the fraction having the highest cetane rating to the lowest. Another point is that the higher the cetane rating of these compounds, the lower the heating value. This is important because it defines the amount of power that can be derived from a given amount of fuel. In general, paraffinic compounds are essential to autoignition at lower temperatures. However, after the engine has warmed up these compounds are not needed because they have lower heating values and have a tendency to produce soot (like the flame from a candle). But for our purposes, the most desirable fuel is one with a low carbon-to-hydrogen ratio. In order words, a fuel high in paraffin compounds. This leads to a higher cetane rating, hence a shorter delay time, which is critical for small, high-speed engines.

Miniature model engine makers have sought ways around this problem for quite some time. Compression ignition engines in this class have succeeded by using different blends rather than ordinary diesel fuel. This is necessary because these engines do not build up enough heat to burn a pure diesel fuel. Most use a blend of kerosene, ether, and castor oil. Model diesel engines do not have compression ratios as high as full size diesel engines. To compensate for a lower compression ratio, ether is added to the fuel because it has a very low ignition temperature (a flash point of -49°F). Ether itself is a very poor fuel, but it readily breaks down into easily burned peroxide compounds with a low ignition temperature which provides localized hot spots that ignite the main fuel (kerosene). Recent reports have indicated that engines in this class can be operated at even lower compression ratios than before through the introduction of a catalyst. The catalyst works by promoting the formation of peroxides formed by the decomposition of ether when it is exposed to the catalyst and hot air.

Another class of model engines that operates very similar to the miniature diesels are the "glow" engines. These engines use an alcohol-based fuel with nitromethane as an additive. The most notable difference between the two engines is that the "glow" engines use a glow plug, hence the name. The glow plug works by acting as a heat reservoir that provides a hot spot for ignition. It also works by providing a catalytic action that enhances the burning process. The filament is made of platinum, which catalyzes a wide range of materials, including nitromethane and diesel fuel.

Research into early diesel-engine development shows that the use of platinum to enhance the burning in diesel engines was widespread before 1900. The progress in the development of injection systems and diesel engines in general has made the use of catalysts in modern engines obsolete. However, the special problems involved in the development of small diesel engines have led us to investigate the possibility of using this technology again.

As the size of a diesel engine decreases, some of the limitations of these engines become quite acute. The injection systems in diesel engines have an upper operating limit of 4000 rpm. Above this speed, the injection becomes very erratic. This is caused by a phenomenon called needle bounce, which causes multiple injections into the cylinder during high speed operation. When this phenomenon is imposed upon small diesel engines the size of the engine becomes limited. Consider a diesel engine designed to operate at the upper limit of 4000 rpm and provide 1.5 brake horsepower (bhp). Assuming a mechanical efficiency of 85% and a stroke-to-bore ratio of 1.25 to 1, the bore of the piston would have to be 1.5 inches and the stroke 1.875 inches. Clearly, this is too large for the purposes for which we intend to use this engine. The introduction of a carburetor would alleviate this problem, but it would also lead to poorer mixing of the fuel.

This is the reason it was decided to look at the model engine field. These engines operate at extremely high speeds which allow them to generate much more power in a smaller package. These engines are also known as hygroscopic engines. A swirl is produced in the fuel-air mixture after it passes through the carburetor, enhancing the mixing process. The main technical hurdle was that these engines are not designed to run on diesel and could not run on diesel unless an additive was used. It was clear that the use of an additive would be unacceptable from a military perspective because of the logistical burden it would impose. It was because of this that the use of a platinum catalyst was investigated. Initial investigations showed promise that this approach could work. The catalyst works by hydrogenating the fuel to produce a better carbon-to-hydrogen ratio. It was thought that the additives could be eliminated if this procedure was successful. The ultimate plan is to introduce a platinum filament inside one of the model engines. The filament would be attached with ceramic insulators to not only act as a catalyst, but also as a localized hot spot that would promote combustion.

Carbureted, Compression-Ignition Engine

The first engine chosen was a PAW brand 0.60 in^3 displacement engine (See Fig. 1). This engine produces approximately 1.5 hp at 7000 rpm. It weighs approximately 22 ounces. This engine was chosen because of its ease of starting and its robust construction. It has a wing nut on top which allows the compression ratio to be changed conveniently. The engine was configured with a flywheel instead of a prop to allow easy access to the controls.

The engine was placed on a test stand and run for approximately one hour using five-minute intervals to break it

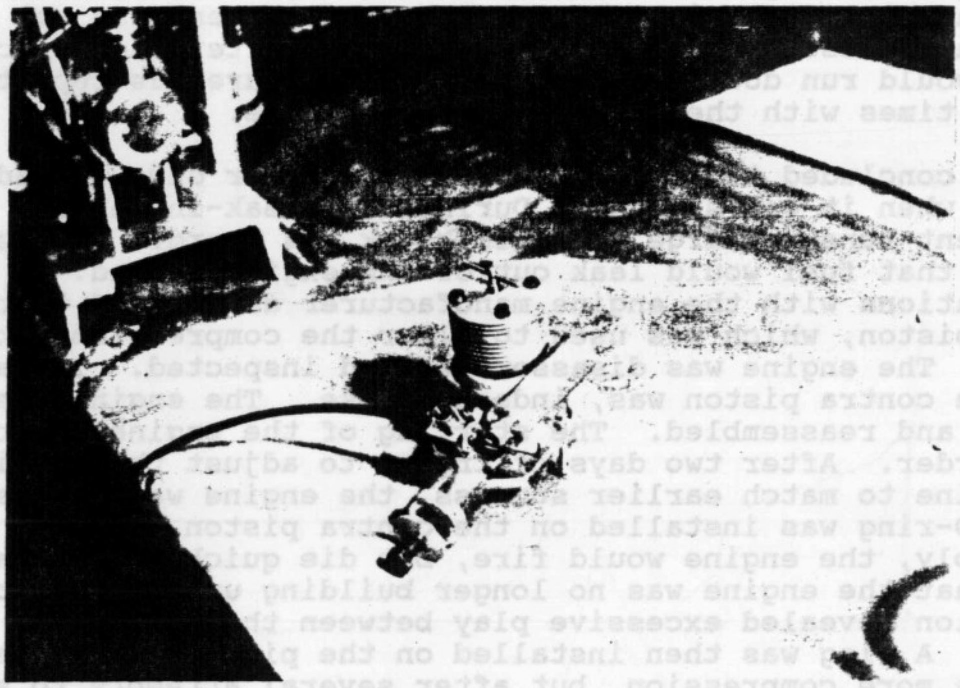

Figure 1
Compression Ignition Engine with Variable Compression Ratio Screw on Cylinder Head (PAW Engine).

in. After the initial break-in period, two flasks were placed on the test stand and connected to allow dual fuels during operation. One tank was filled with diesel fuel and castor oil, while the other was filled with the model blend of kerosene, ether, and castor oil. The engine proved to be easy to start, often requiring little more than a sharp flip of the flywheel. The engine was started using the model blend. After the engine had warmed up, the diesel fuel line was opened to allow a mixture of both fuels to enter into the engine. Engine rpm's increased considerably, but engine performance also became noticeably smoother. The engine was allowed to run for about a minute, using the dual fuel mix, and then the model blend was cut off to allow only diesel fuel. Some five seconds after the switch, the engine would run down and stop. This procedure was repeated several times with the same results each time.

We concluded the engine needed less ether once it had warmed up than when it was started. During the break-in period, and in subsequent runs, a large metallic knock was heard. It was also noticed that fuel would leak out of the engine's head. Consultations with the engine manufacturer determined that the contra piston, which was used to alter the compression ratio, was faulty. The engine was disassembled and inspected. It was found that the contra piston was, indeed, loose. The engine was cleaned and reassembled. The starting of the engine was becoming much harder. After two days of trying to adjust the settings of the engine to match earlier success, the engine was disassembled and an O-ring was installed on the contra piston. Upon reassembly, the engine would fire, but die quickly. It was found that the engine was no longer building up compression. Inspection revealed excessive play between the piston and the sleeve. A ring was then installed on the piston. This seemed to generate more compression, but after several attempts to start the engine, it was noticed that the compression was again very weak. Disassembly showed that a piece of the ring had sheared off.

At this time, a new engine equipped with rings was purchased. This engine was an O/S Model 108 (See Fig. 2) which used a glow plug. This engine was much larger than the previous model, 1.08 cubic inch versus 0.60 cubic inch. It was capable of three horsepower at 9000 rpm. It was thought that the lower surface-to-volume ratio of the new engine would enable this engine to sustain combustion with diesel fuel. A head that allowed for variation in compression was installed on the engine. An attempt was then made to start the engine. This was not successful. The original head was then reinstalled.

This engine was designed to run on the alcohol mix. The engine was allowed a break-in period of several hours using an alcohol/nitromethane mixture. The higher-compression head was then reinstalled and an attempt to run the engine on the

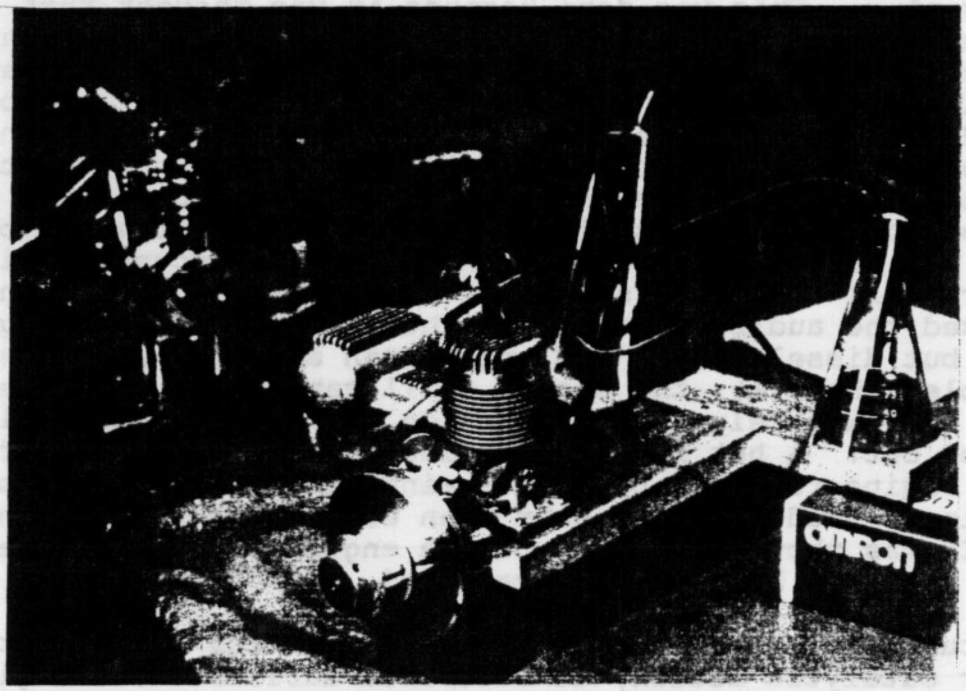

Figure 2
O/S Brand Engine with Glow Plug Ignition
Running on Nitromethane and Diesel Fuel.

ether/kerosene blend was again made. Once again the engine would not run. The glow head was then reinstalled. The engine was started and allowed to warm up. After the warm-up period the fuel was switched from the alcohol blend to the kerosene blend. The engine ran, but it was very rough and a loud metallic knock was heard. The engine was shut down and the kerosene blend was changed to a diesel/castor oil mix. The engine was again started on the alcohol blend and then switched over to the diesel. The engine ran for approximately 25 minutes while on the diesel mix. The performance was very rough, and again the metallic knock was heard.

A variable resistor was installed between the battery and the glow plug. This was done because it was thought that the fuel was igniting much sooner than it should, which was causing the knock. The engine was restarted and switched to diesel fuel. The resistor was then adjusted to try to eliminate the knock. This did not work. Instead of reducing the knock, the engine just died out. Upon further investigation, it was found that the knock was caused by too much castor oil in the fuel. The port design of this engine was different from that of the PAW brand. Because the ports were smaller, the oil would accumulate in the cylinder until there was so much it filled the compression space and caused the audible knock. Another run was tried using nothing but diesel fuel. The engine ran and the knock was less noticeable. However, the engine still ran roughly. The engine ran for approximately five minutes on straight diesel fuel and then died out. A hot start with diesel was tried successfully and the engine ran for about five minutes. After this procedure, the engine seemed to lack power when used under the original fuel. It was thought that maybe the engine sleeve had been scored, but this has not been verified.

Future tests should revolve around placing a filament in the head for catalytic action, as described earlier. The engine should be modified to revolve around port design and compression ratio variation. Dynamometer tests should also be performed to determine the power curve. Different types of carburetors should be tried to improve the fuel economy of the engines. The fuel economy of these engines was measured to be roughly one liter per hour. Overall, the outlook is favorable and these engines can be modified to run on diesel fuel.

Direct Fuel Injected, Spark Assist Ignition Engine

In an effort to speed development of this miniature diesel program, the advantages of a simple loop scavenge two-cycle engine became apparent. Commercial industry has predominately adopted this type for the engine sizes applicable for this project, thereby providing development support basis. However, from a technical viewpoint, the miniature two-stroke, when converted to diesel operation, has the following drawbacks:

- Necessarily high surface/volume ratio
- Low cylinder scavenge/volumetric efficiency
- Demands relatively high injection pump speeds

The primary development objective was to overcome these deficiencies by careful design and development of the fuel injection and combustion system. To appreciate the level of achievement necessary to meet the current requirements, a quick review of the cycle efficiency will explain the fundamental technical barriers.

Cycle Efficiency

The overall efficiency depends primarily on the ratio of expansion. Although higher maximum pressures increase this ratio, they necessitate heavier dynamic parts, along with higher seal loadings and therefore higher frictional losses. Consequently, for equal mean effective pressure, the cycle which yields its efficiency at the lowest peak pressure will always show to advantage in practice.

The thermal efficiency of an engine is comparative on the basis of its direct heat loss by interchange of heat between the gases and the chamber. The actual extent of the loss cannot be completely accounted for because it depends on many variables. The more predominate factors are:

- Size and speed of the engine
- Form and area of the combustion chamber (Surface/Volume)
- Degree of turbulence or air-swirl

In the case of very small engines of the size relative to this development, the bulk of the direct heat losses are larger because of their greater surface/volume ratio which are inversely proportional to displacement. Unlike the petrol engine, the true diesel engine inherently has operational speed limitations which otherwise aid in reducing heat loss. This is because the compression ignition engine handles the entire job of injecting, atomizing, and igniting the fuel inside the cylinder in only a few thousands of a second. Therefore, conditions essential for good combustion must be provided from the fuel injection to atomize the fuel, and the combustion chamber design to mix the air with the fuel in order to provide spontaneous combustion. Additional swirl and turbulence tend to speed the combustion process. However, this also slightly increases heat loss. Increasing the compression ratio will likewise help to increase the thermal efficiency, but only at the relatively greater expansion of higher frictional losses from the internal pressure expanding the rings. The fact is, the very minimum ratio of compression necessary to sustain compression ignition of a miniature diesel engine is always above, rather than below, the optimum in the interest of mechanical efficiency.

Direct Fuel Injection

In the case of the miniature diesel engine, the fuel injection system design presents unique difficulties with regard to fuel metering from the corresponding small pump size. Consequently, the downsizing of a typical industry jerk pump places higher demands on plunger and injector valve tolerances requiring l pped fits within millionths of an inch. The design objective t us demands relatively low-cost component manufacturi..g with the capability of precise metering.

The design approach taken was to adopt as much off-the-shelf equipment and modify it to fit our miniature engine application. Early experimentation and calibration testing of these industry-available pumps were not capable of metering fuel quantities down to the volumes required for our application (i.e. 3 mm^3 per stroke maximum). Rack settings next to the cutoff point of the plunger proved to deliver unreliable fuel quantities that were too sensitive for any practical testing.

The immediate development solution to this problem was to incorporate a simple needle valve between the plunger and the delivery valve. The excess fuel is then returned to the fuel tank. This provided a means of bypassing the excess fuel delivery that can be accurately calibrated on the test bench.

This method thus allowed the pump to operate at its designed capacity while restoring the helical rack metering sensitivity to practical governing adjustment standards.

The collaborative design effort of this injection system resulted in a specialized 90-degree unit injector as the prototype. The direct combustion chamber design dictated the need for a multiorifice injector which, in this case, is two orifices of just 0.005 in. diameter each. The pintle valve preload was tested between 2700 to 3000 psi. The pressure was directly proportional to atomization. However, reliability of fuel delivery and metering proved to be very unstable at higher opening pressures and operating speeds. The increased pump entry pressure with the addition of a check valve helped stabilize the injection volume, but still has inconsistencies in spray patterns determining atomization versus penetration. It appeared the higher pressures reacted more favorably toward increased penetration or spray length which, unfortunately, proved to be excessive, resulting in fuel impingement on the combustion chamber wall.

Overall, the performance of the common jerk pump can be improved and specifically scaled to meet these minute fuel delivery requirements. However, it has become apparent that the relatively small amount of fuel we are attempting to meter approaches that minute volume considered to be compressible.

Therefore, the systems were on the verge of being elastic and rather unpredictable in performance. The development direction here would be to incorporate a smaller plunger bore to stroke ratio that tends to become prohibitive in manufacturing tolerance.

Combustion System

The combustion system employed in this engine may briefly be described as a two-phase divided chamber. The geometry is rather simple (See Fig. 3).

Attached to the piston is a cylindrically-shaped part called the poker. Its upper face is part of a toroid. Its periphery is vertically slotted and aligned with the fuel injection spray pattern. The poker engages [begins at 32° Before Top Dead Center (BTDC)] a recess in the cylinder head which has a cylindrical lower section and a one-half toroidal upper section. In the top center of the toroid, the injection nozzle is mounted so that its spray tip just extends into the chamber. Each injection orifice is directed at the upper end of the poker slot. Advantages of the chamber are numerous:

- Peak cylinder pressure is moderate
- Knock intensity is low
- Fuel sensitivity is broad

The objective here is the development of a combustion system capable of delivering the maximum power with a minimum peak pressure yielding an overall lighter engine construction. The prototype also incorporates a spark plug located in-line with one of the injection sprays. This was intended to aid in the early development of the engine and would later be replaced by a glow plug once the timing and rate of injection had been established. (i.e. The engine that reveals the nearest peak to mean effective pressure will always show to advantage.)

Preliminary Test Results

The early approach of modifying an existing commercial petrol burning engine, Quatra 42 CDI (See Fig. 4), proved to be unforgiving for the purposes of prototype diesel combustion analysis and development. Early operation of this modified test unit (See Figs. 5, 6, and 7) proved to lack the structural integrity necessary for diesel combustion development. Initial testing proved to be somewhat of a flash in the pan, resulting in excessive time and effort spent on modifying and calibrating the injection system. As it turned out, the real problem was that

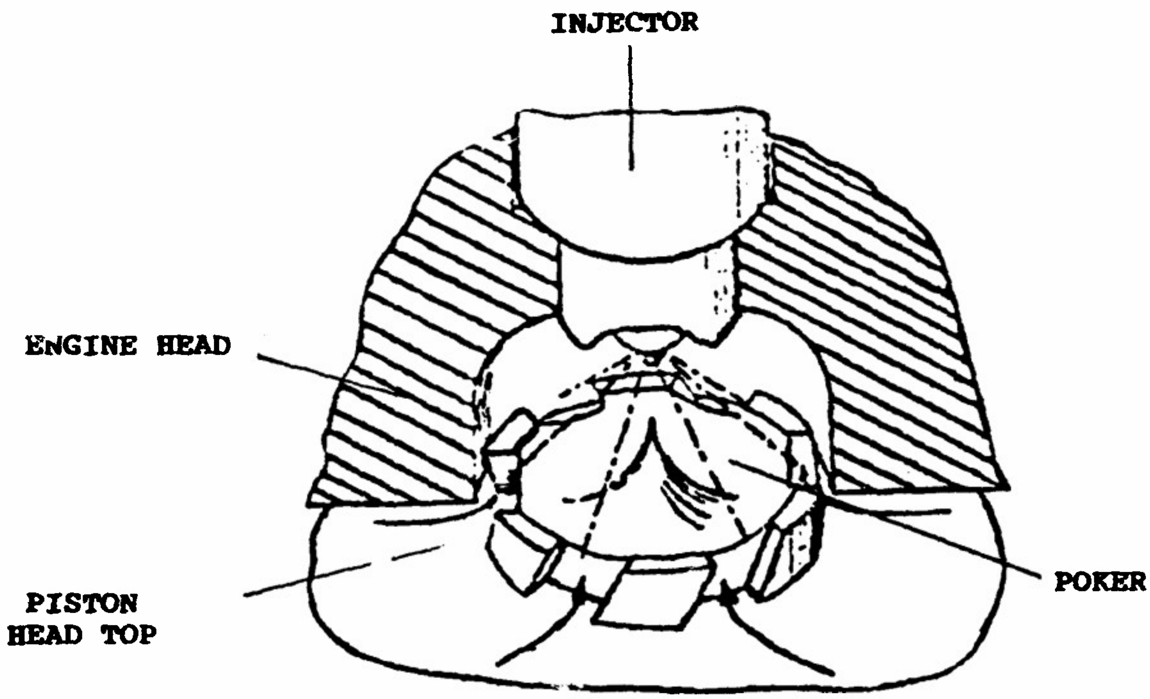

Figure 3
Direct Fuel Injection Combusiton System.

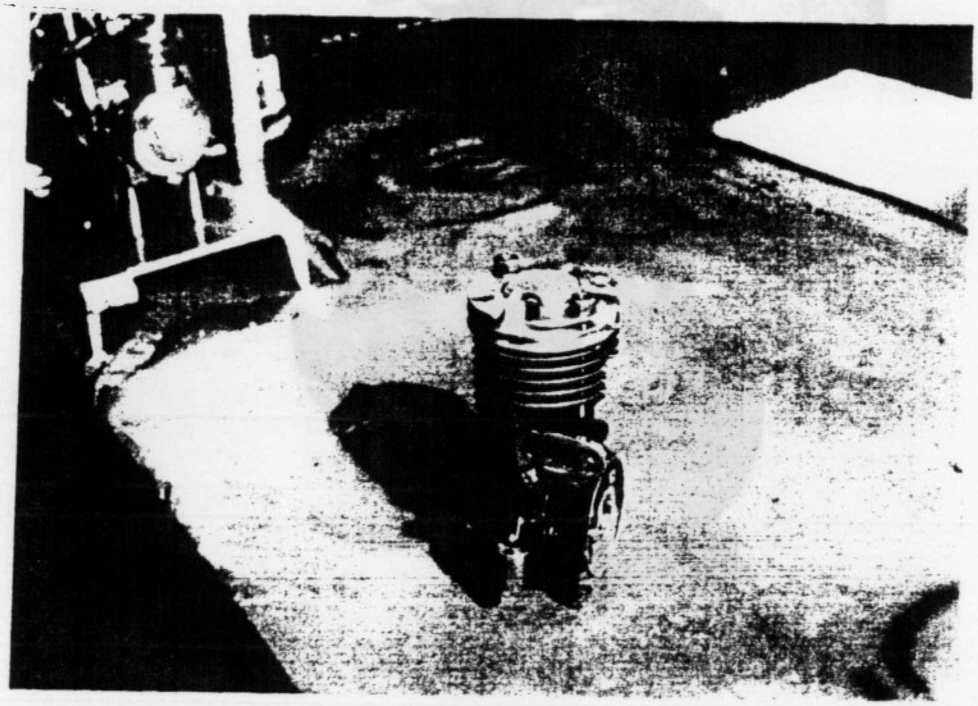

Figure 4
Quatra 42 CDI Engine.

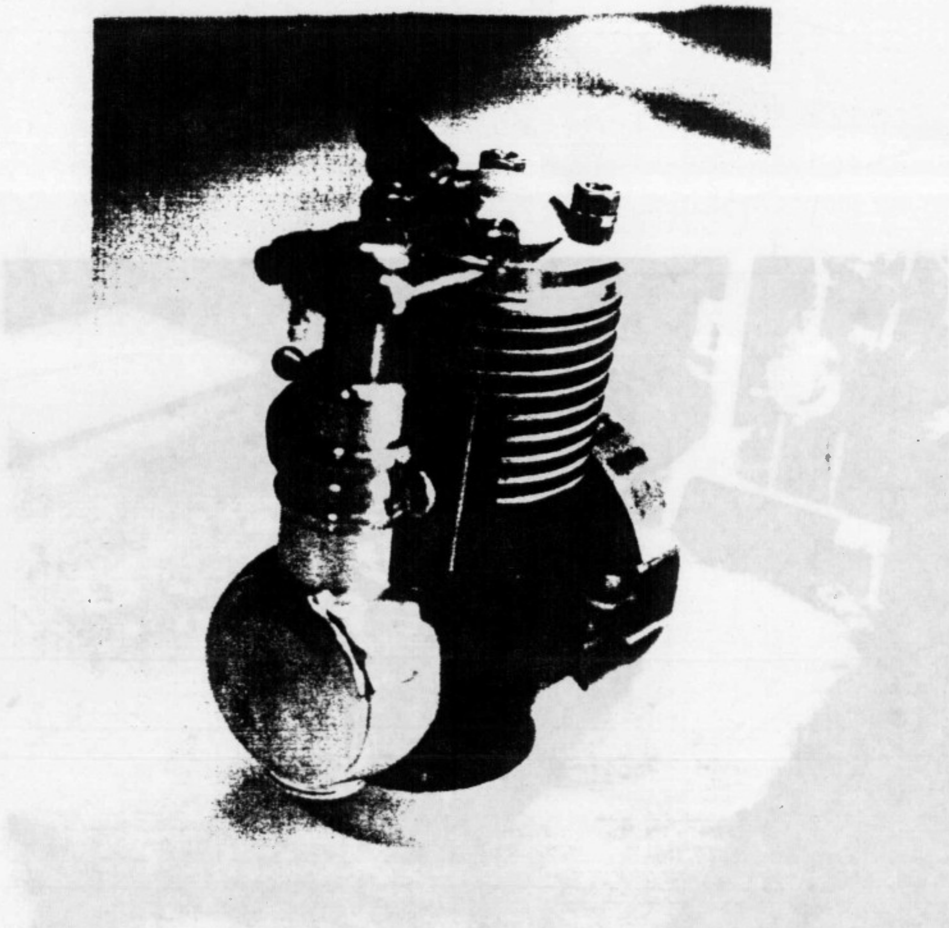

Figure 5
Modified Quatra 42 Engine.

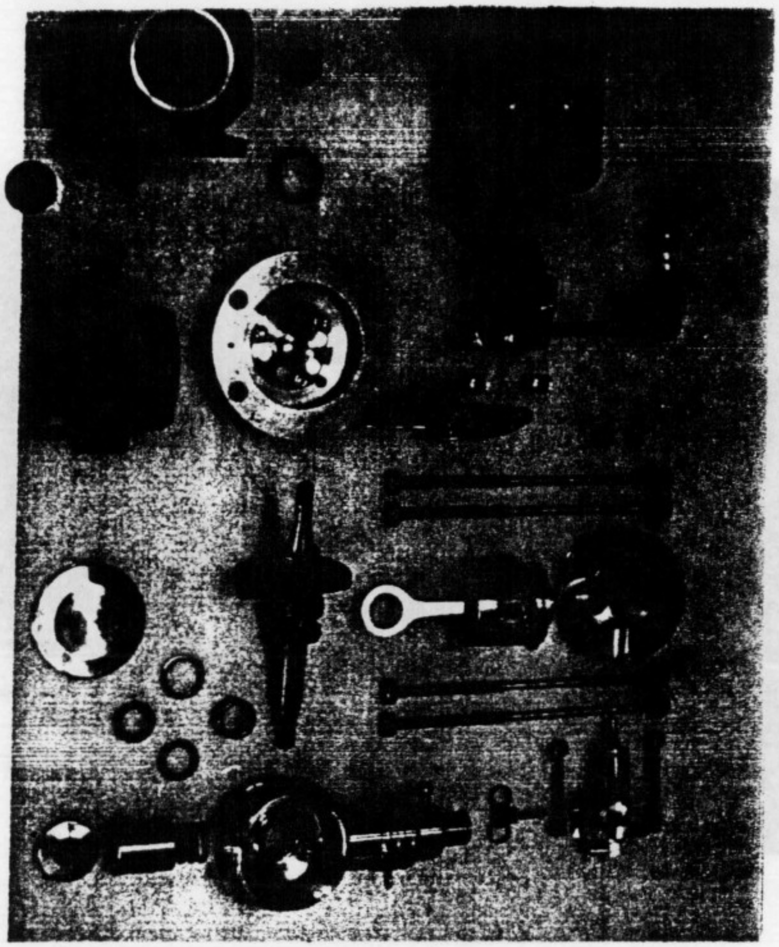

Figure 6
AMI Engine Internal Components.

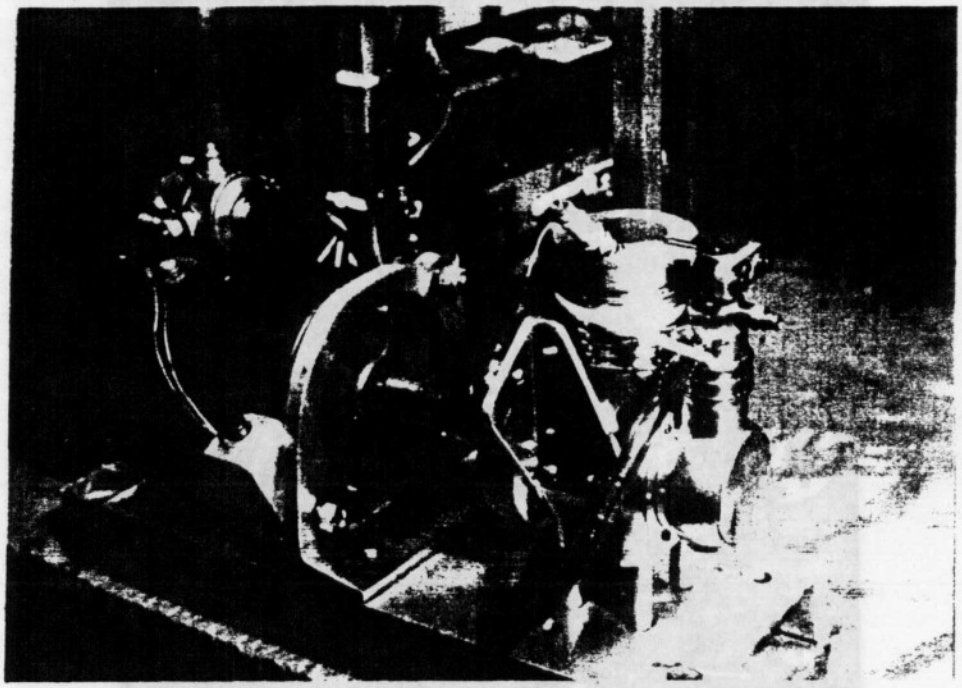

Figure 7
AMI Spark-Assisted, Fuel-Injected
Engine with a Laboratory Starter Motor.

the cylinder lost its shape due to the excess amount of pressure induced from the newly incorporated cylinder studs, resulting in an increase of the bore by 0.005 inch in the center, like a barrel.

This dictated the need for a solid test engine core that was immediately constructed with all the emphasis placed on structural integrity and repeatable performance.

Although, to date, the engine (See Fig. 7) has not completed any long, sustained test runs, it has been successfully started and operated for short periods on numerous occasions. This engine has proven to represent a good starting opportunity to complete the development of a proper combustion system with the proper timing and rate of fuel injection.

Fuel Injected, Compression Ignition (Cold Start) Engine

Fuel Injection

This engine (See Fig. 8) was designed to allow for variable injection variable volume by adjusting the Yanmar injection plunger pump's stroke. The injector pump was set up on a slide secured by socket-head cap screws. The slide included the injector pump and tappet to avoid side loading of the injector pump plunger. The stroke of the plunger pump was varied by moving the slide's position in respect to the cam lobe located on the crankshaft.

The other varying factor was the rack on the plunger itself. Together, with the varying of plunger stroke and varying of rack angle, small amounts of fuel might be produced.

Air Intake System

The engine's air intake used a rotary valve in the crankshaft similar to what is used in model-airplane engines. This was used over a flapper valve because it would have longer opening duration, thus more air.

The air enters the crankcase and is then compressed on the piston's down stroke. The air is then released in the cylinder through six helix-angled passages. The helix angle of the passages was 15 degrees. This was intended to produce scarification and swirl in the intake air.

On the compression stroke, the swirling air is compressed into the toroidal clearance volume. The injection then occurs at approximately 10° BTDC into the swirling air.

Test A

The engine was intended to start with a pull rope. The rope

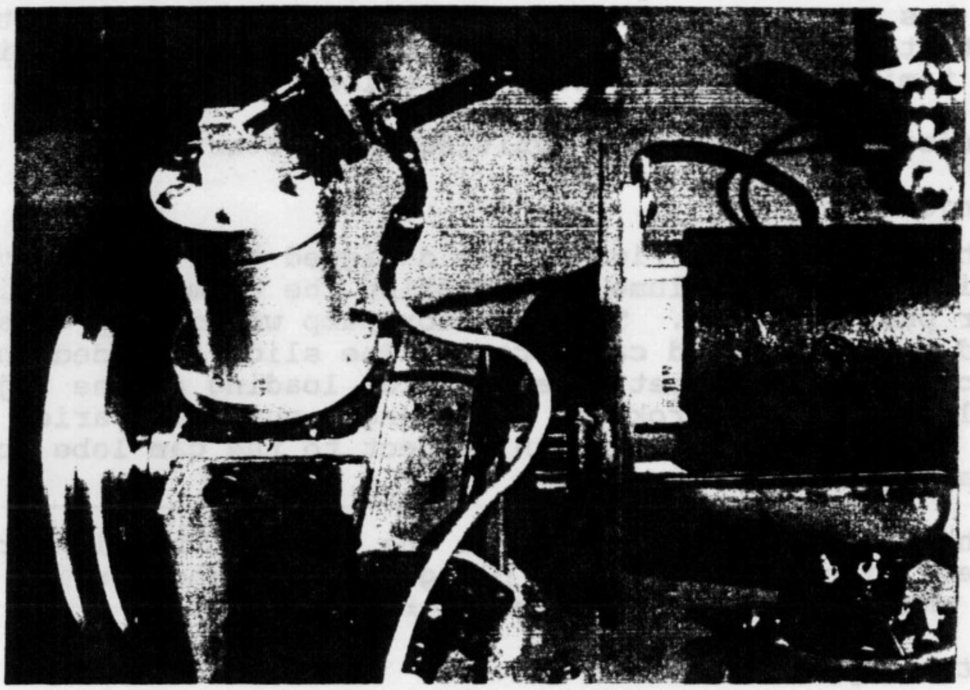

Figure 8
AMI Compression-Ignition, Fuel-Injected Engine with Laboratory Starter Motor.

was wrapped on to the flywheel. The engine was pulled many times in this fashion while the injection timing and injection volume were being varied. It was discovered that "blowby" past the compression rings was occurring. Even very small amounts of "blowby" greatly reduced the possibility of compression ignition.

The cylinder sleeve was then replaced. The new sleeve and piston had a total clearance of 0.00075 inches. This new arrangement was better, but "blowby" was still evident.

Test B

The test was continued. Ether was also employed. With the ether, several ignitions occurred on an ether and diesel combination. The test was conducted when the ambient temperature was 64° F. Without the ether, an audible knocking occurred on straight diesel and on occasion small amounts of smoke were visible. It could not be determined whether the smoke was actually combustion products or distillates of diesel. However, the smoke had a noticeably different smell.

Even with ether, we could not get sustained combustion. It was felt that the diesel fuel was not being atomized into small enough droplets through the 0.008-inch nozzle. It appeared that some of the smallest droplets were combusting, but they did not start a chain reaction necessary for running. It was also felt that too much heat transfer from the combustion was occurring during the pull-cranking operation and that no heat was being consumed between cranks. It appeared that hand cranking did not produce high enough speeds to produce adiabatic compression. During the first two tests, several crankshaft failures occurred. It was determined that the press fits were not adequate for the high pressures of diesel engines. The crankshaft would go out of alignment during cranking. Several set screws were used as alignment pins. This was barely satisfactory.

It was determined that more cranking speed was needed. It was decided that an electric starter should be used for cranking. A standard Ford starter was used (See Fig. 8).

Test C

Test C began by using the electric starter. The engine was turned over at a higher rate of speed by the electric starter. After 150 to 250 continuous revolutions, several diesel ignitions occurred; however, they did not occur on consecutive revolutions. This high rpm consumed and increased the heat of the combustion chamber.

During this test, the heat expanded the aluminum piston more quickly than the steel sleeve and an interference occurred, seizing the piston to the cylinder wall. This irreparably damaged the piston rings and sleeve, thus stopping the test.

This concluded the test of the pure compression diesel engine, pending more funding for further development.

The fact that ignition was achieved, even though sustained combustion did not occur, indicated that the engine configuration is headed in the correct direction. It is believed that sustained combustion and operation did not occur, mainly because an uncontrolled surplus of fuel was injected into the system and thereby acted as a large heat sink. The volume of injected fuel in such a low range is very difficult to set and control even using only one 0.008-inch id injection hole. Our experience suggests that a carburetor system may offer more potential in achieving proper nebulized fuel volumes without creating an excess fuel heat sink. AMI would like to continue development work in this direction on this engine. With proper fuel metering and with the inherent high compression ratio of the engine, sustained ignition and operation have a good chance of being achieved. AMI believes further work on this engine should be pursued simultaneously with further work on the spark-assist unit.

Conclusions and Recommendations

1. In terms of priorities, the spark-assist approach should be the main focus of further work. While the glow plug approach holds promise, it carries with it the penalty of requiring an ever present battery. A successful mastery of the spark-assist model would be a major step toward the ultimate objective.

2. The fuel feed system in the short term should be carburetion. While a finely controlled fuel injector system may eventually become achievable for such small injection volumes, a long development path lies ahead. AMI believes that a carburetor can be successfully designed and fitted to the spark-assist model.

3. The compression ratio of the current spark-assist model can be increased with better piston ring design. AMI believes the current 18 to 1 ratio can be increased to 23 to 1.

4. The current true diesel model should be repaired and also converted to a carbureted, spark-assist model so that it can serve as a parallel test unit.

5. Of lesser priority would be continued work on the glow plug, high compression engine operation with pure diesel fuel. Given a willingness to accept the logistics burden of battery supply, this approach could possibly reach a field operational status quicker than any other. AMI has already worked out the details of fitting a glow plug to a variable compression head, and would like to proceed on experiments with this configuration.

This document reports research undertaken at the U.S. Army Natick Research, Development and Engineering Center and has been assigned No. NATICK/TR-93/139 in the series of reports approved for publication.

APPENDICES

APPENDIX A

Fuel Injection and Compression Ignition Details for the Cold-Start Minidiesel Engine

Appendix A

FUEL INJECTION AND COMPRESSION IGNITION DETAILS FOR THE COLD-START MINIDIESEL ENGINE

Bore	1.573 in
Stroke	1.500 in
Effective Stroke	
(From top of exhaust port)	1.0625 in
Angle of exhaust port opening	148°
Angle of intake port opening	126.5°
Piston area	1.943 in^2
Displacement	2.9145 in^3
Effective Displacement	2.0644 in^3
Piston to cylinder top clearance	0.005 in
Clearance volume port (1)	
(Volume of port head clearance)	0.0100 in^3
Clearance volume port (2)	
(Torrus volume in head and piston)	0.0723 in^3
Total clearance volume	0.0820 in^3

Compression ratio (effective)

$$\frac{2.0644 + 0.0820}{0.0820} = 26.18 \text{ to } 1$$

Piston:

Piston diameter	1.572 in ± 0.0005 in

Compression Ring 4:
(From an "ECHO" 40 mm diameter piston)

Ring thickness	0.049 in ± 0.0005 in
Ring groove width	0.050 in ± 0.001 in
Ring groove land (distance between rings)	0.080 in ± 0.001 in

A CONTINUED

Deck height	1.0473 in
Wrist pin diameter (hollow)	0.375 in ± 0.0002 in
Length	1.250 in ± 0.005 in
Wrist pin bearings 2 req (needles) 0.5625 of; 375 icl; .3125 wide	

Connecting rod:	
Wrist pin diameter	0.375 in
Crankshaft end bore	0.750 in
Crankshaft bearing	0.750 in
Needle bearing	0.500 in
Length	0.4375 in
Rod length	2.500 in
Rod ratio $\frac{2400}{1.50}$	1.667 in

Crankshaft:	
(Fabricated from individual pieces pressed together with 0.002" press fits)	
Main journal diameter	0.750 in
(1) Air intake side	0.500 in
(Flywheel side)	0.750 in
(2) Injector side	0.500 in
Output shaft	
(1) Flywheel side 1/8" key	0.500 in dia.
(2) Injector side	0.500 in dia.
Connecting rod bearing	0.500 in dia.
Needle bearings	0.4375 in long
2 Flyweights, throw diameter	2.25 in

A CONTINUED

Flywheel
Diameter 6.0 in
Weight, approximate 1.3 in
Approximate moment of inertia 7.0 lb in^2

Injector:
Cam: standard "Yanmar" injector cam lobe
Injection angle: variable
Approximate 10° BTDC
Injection duration at low speeds
Approximate 6.5° (depending plunger pump stroke)

Injector: standard GM pencil injector from 1970s GM diesel cars. It uses a pentle valve mechanism with variable pentle valve opening distance. The nozzle was replaced (the original nozzle had four 0.008 in diameter holes) with a nozzle with a single 0.0083 in diameter hole.

Injector pump Yanmar YPFE-M fuel injection pump.

APPENDIX B

Direct Fuel Injection and Spark Assist Ignition Engine Details

Appendix B

DIRECT FUEL INJECTION AND SPARK ASSIST IGNITION ENGINE DETAILS

Bore = 40 mm
Stroke = 34 mm
Swept Volume = 42 cc
Effective Stroke = 23.75 mm
Surface/Volume = 0.42 (Effective)
Effective Displacement = 29.85 cc
Clearance Volume = 1.8 cc
Compression Ratio = 18:1
Connecting Rod Length = 60.25 mm
Exhaust Port Timing = EXOP 75° BBDC
Intake Port Timing = INOP 58° BBDC
Fuel Pump Plunger Diameter = 5 mm
Fuel Pump Stroke = 6 mm